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EER Improvement on a Reciprocating Hermetic Compressor

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INTRODUCTION

The EER compressor improvement,whether introduced in a heat pump cycle , or in other more traditional applications,such as air-conditioning and refrigeration , represents one of the most important factors for the efficiency increase in general and for widespread of the energy saving applications.

As a specialized producer of hermetic compressors since a long time Aspera has faced efficiency problem with significant results. Two years ago the C.N.R. (National Research Council) , appreciating Aspera's activities in this field and in the picture of finalized energy program , has offered our Company a three years contract specifically for the EER improvement of heat pump hermetic compressors . (Research contract number 78.01187.92 ; title "Improvement of EER in hermetic motorcompressors for heat pumps"). Such contract has to be considered the first step of the heat pump project and the relevant target is,as far as 5800 Watts (Ashrae-T) hermetic compressor,a minimum improvement of 20% versus average actual values reached in 1978.At the same time it has been required the best economical target in the industrial cost of the product.

GENERAL RESEARCH LINES AND METHODS USED

The present action taken on compressor can be summarized with the following research lines (RL.):

1) Modifications to the mechanics and fluodynamics components(ducts,valves,suction and discharge ports)

2) Electric motor speed reduction (from 2 to 4 poles) and corresponding displacement increase

3) reduction of the heat transfer to limit the suction gas overheating

4) Combination of RL.3 with RL.2 solutions and making prototypes for practical research on heat pump applications

Generally it can be observed that the reciprocal influence between the factors which govern the operation of a hermetic compressor is such that the modifications introduced for electrical and mechanical optimization are not always followed by an overall efficiency improvement.

This creates the need to accurately test every modification in order to evaluate the single effect with considerable theoretical and experimental work.

In particular,to reduce the development periods,for hypothesis involving improvement of the thermodynamic cycle , a mathematical simulation was introduced.

This was developed as follows :

1) Thermodynamic simulation based on adaptation of the 'Compressor Simulation Program with Gas Pulsations' applied on a multicylinder compressor with correspondent resonant volumes /1/

2) Program elaborated by Aspera which calculates the performance of an electric motor in the normal operating 'warm' conditions

The above two steps interactions are described in appendix in which is considered the heat transfer between the electric motor and the working fluid.

A S P E R A ENG. R & D - dept.		EXPERIMENT NUMBER															
		RL.1					RL.2			RL.3				RL.4			
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
MODIFICATIONS	FREQUENCY								***							***	***
	POLES NUMBER							***	***					***	***	***	***
	VALVE PLATE		***	***	***	***						***	***				
	VALVE LEAVES			***													
	SUC. LEAF STOP					***											
	DISCHARGE SECTIONS				***	***											
	DIRECT SUCTION									**	***	***	***	***	***	***	***
	OIL COOLER												***		***		***
	DISPLACEMENT						***	***	***					***	***	***	***

Table 1 - *Synoptic description of the action per groups of tests*

Standard condition : 50 Hz frequency ; 2 poles
Modified condition (***) : 60 Hz frequency ; 4 poles ; displacement about +70 %

DESCRIPTION OF EXPERIMENTAL RESEARCH

A standard calorimetric room is the main facility used in this research together with standard electric motor brakes and equipment for analysis of lubrication circuits.

All the calorimetric measures have been made at the following conditions :

- Suction and ambient T = 35°C (308°K)
- Condensing T = 55°C (328°K)
- Evaporating T = 7,22°C (280°K)
- Liquid T = 47°C (320°K)

Table 1 reports the action taken in the single Research Lines.

Research Line 1 - (RL.1) -

The aim of this research line is to investigate on possible compressor improvement without changing the design, mantaining the mechanical (stroke, bore, ducts etc.) and electrical parameters.

In this phase we have worked on the valve plate , suction and discharge ports , on valve leaves , leaves stop and on the discharge piping section.

Regarding the discharge ports, it was confirmed the existance of an optimum section value which compensate the efficiency reduction due to the increase of the dead space./2/

Such effect, also confirmed by mathematical simulation , is of lower importance when compared to an increase of the suction ports sections where the gas velocity reaches Mach number values higher than the ones on the discharge ports.(see fig. 1)

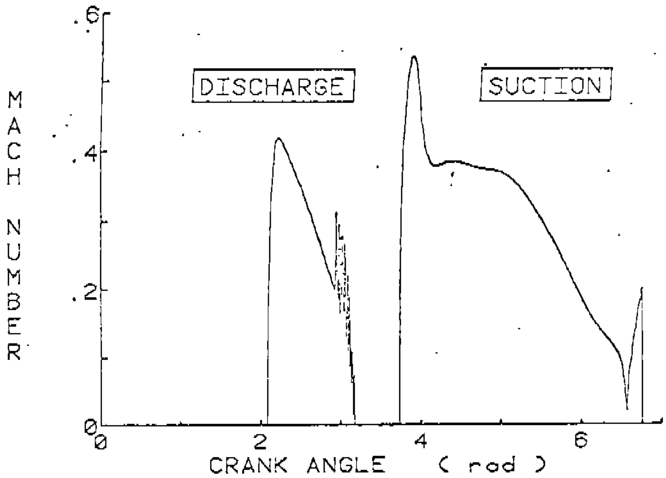

fig.1 - Mach number diagram for standard compressor.

Replacement of suction valve leaves with others of different stiffness has not given a sensible improvement thus confirming the optimization of the standard ones.

In analogous manner the lowering of the suction limiter, obtained in a cylinder cavity, has not given positive results .
The dead space reduction has not given improvements because cancelled by an insufficient flow area . Vibration modes over the first order have had scarse influence in consideration of an enough rigid valve leaf able to whithstand the stress of large suc-

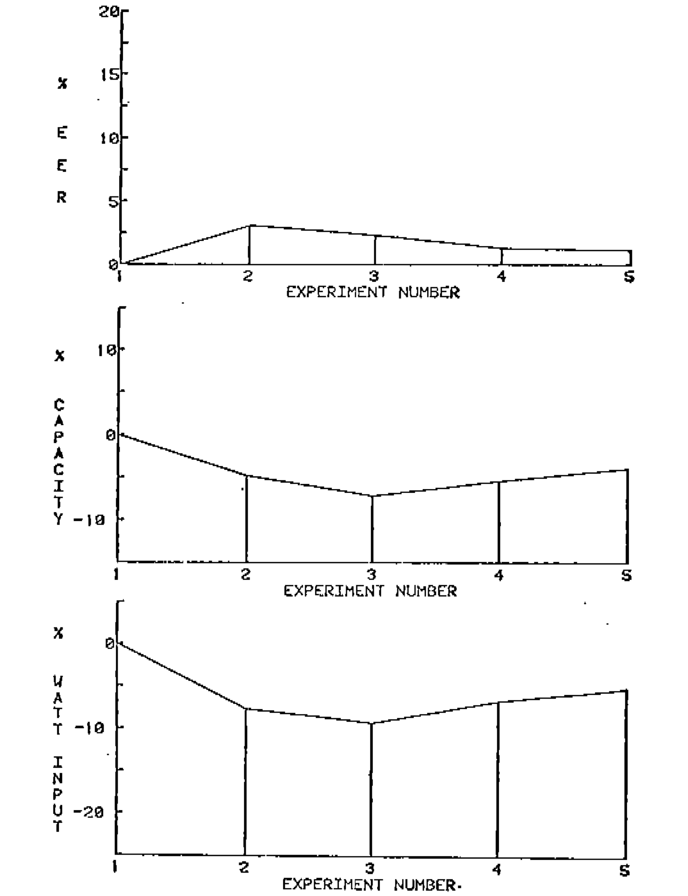

fig.2 - RL.1 - Percentage diagram of results

tion sections.

Action on the intake coils to increase the inside section have not given any significant result.

In the area of this research line , with the aforesaid basic restrictions, the EER percentage improvement was not considerable(3%).

Research Line 2 - (RL.2) -

The EER compressor improvement acting on the rotation speed is analized below by the following considerations.

First consideration concerns performance of the typical open type compressors(see fig.3) which the EER increases upon reduction of the rotation speed.

Second consideration is based on a theoretical approach.The suction and discharge overpressure ΔP,noticeably influencing the efficiency of compressors having automatic valves are supported by formula /3/

$$\Delta P = \frac{\pi^2}{2}\cdot\sqrt[3]{9\cdot\frac{1+K}{\theta^4}\cdot\left(\frac{D}{C}\right)^4\frac{1}{Vg^2}}\cdot\left(\frac{G}{ls}\right)^{2/3}\cdot\frac{u^2}{P}$$

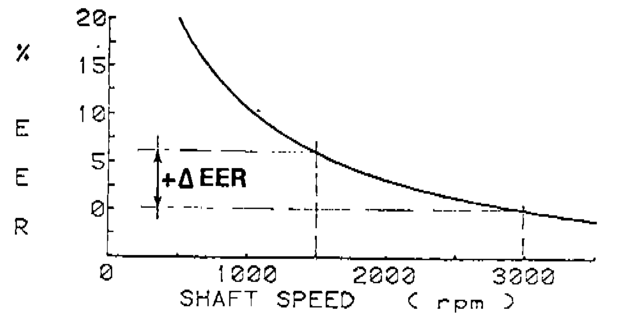

fig.3 - Typical diagram for an open compressor

where:

- p = pressure
- k = friction loss coefficient
- θ = valve leaf opening crank angle
- D = bore
- C = stroke
- V = specific volume
- g = gravity constant
- G = valve leaf weight
- l = gas port perimeter
- s = pressure area
- u = average piston velocity

It can be seen how ΔP depends in a quadratic manner from the average piston velocity 'u' and then by the square of revolution speed.

On the basis of these considerations the performance of a 4 pole motorcompressor has been evaluated.

In order to keep the refrigerating capacity of 5800 Watts an increased displacement compressor was used (about 70% more than the 2 pole version). To compensate the lack of displacement,tests were also made by increasing the revolution speed feeding the motor of 60 Hz instead of 50 Hz.

In every case in this increased displacement compressor we have not modified the valve plates , leaves etc. considering them not a critical part for this type of compressor.

The tests in these conditions have given an EER improvement of 2% against a foreseen thermodynamic simulation increase of 8% . This fact is due to a lower efficiency of the 4 pole motor used in the compressor with the above mentioned stack limitations.

This kind of efficiency reduction in a hermetic motorcompressor is amplified because of the temperature increase of the refrigerant fluid that cools the motor before entering into the cylinder /4/.In fact , in the working point the 4 pole motor has given a lower efficiency of about 5% compared to the 2 pole motor.

Confirmation of this was found in the fol-

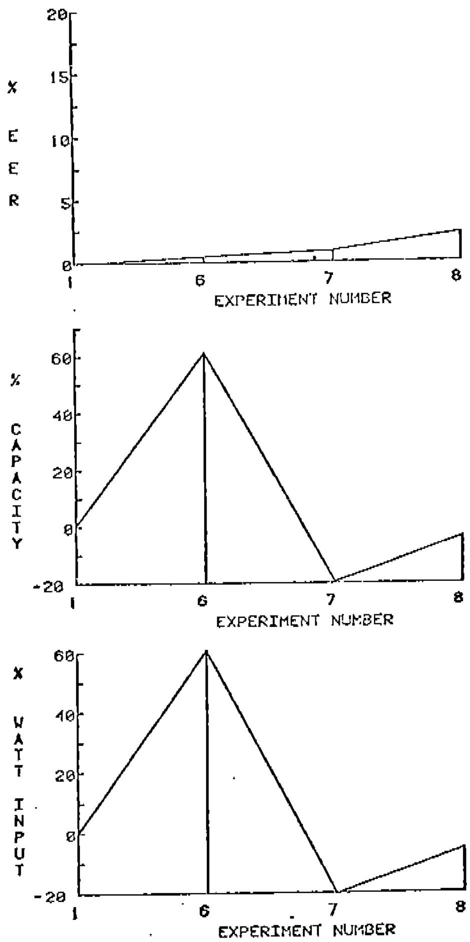

fig.4 - RL.2 - Percentage diagram of results

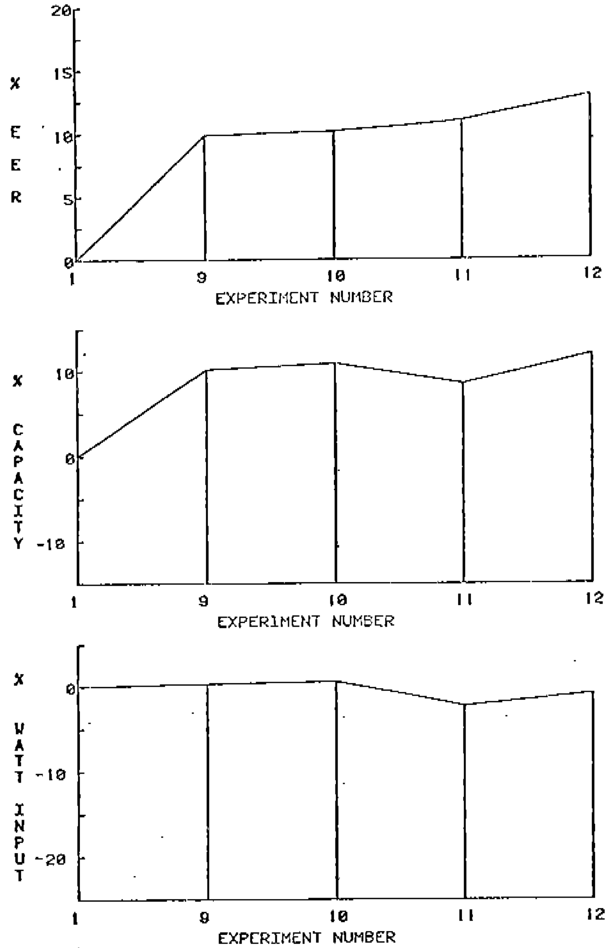

fig.5 - RL.3 - Percentage diagram of results

lowing Research Line 4 where the motor cooling occurs independently from the suction refrigerant fluid.

Research Line 3 (RL.3) -

The concept of this research line is to reduce overheating of the suction gas in the cylinder /5/.

Thus another prototype was assembled with increased shell dimensions having the suction muffler directly connected by a flexible tube with the compressor's outside suction line (see fig.7)

In order to avoid the increase of the internal pressure a hole was practiced on the internal suction line thus mantaining the inlet pressure in the shell.

The use of a hermetic compressor on a system with such a solution requires the use of a liquid and oil accumulator which is normally used in a heat pump system.

In this research line and in the following one the discharge muffler was eliminated.The result of this change was evaluated with an EER increase of about 1%.

Electric motor cooling was guaranteed by internal radiator and by lubrication oil.

A water oil cooler was used and the oil charge was increased to guarantee a higher heat transfer.

Furthermore, the most favorable line 1 solutions have been adopted without using the oil cooler; the tests have always confirmed the results previously obtained.

The final overall increase was of about 11%. On this last prototype the introduction of the water oil cooler has increased of about 13% the EER.

In tests with oil cooler the water circulation power was not taken in consideration; the reason was that such heat elimination method was not necessary.For example in the hypothesis of an air-water or water-water heat pump such power does not influence the system's efficiency.

Research Line 4 - (RL.4) -

The increased displacement compressor prototype with the 4 pole motor previously described was assembled with direct suction.

The input frequency is, as previously mentioned, both at 50 and 60 Hz with and without oil cooler.

The type of electric motor cooling with oil instead of refrigerant is more efficient at higher revolution speed and justify the difference between 50 and 60 Hz results.

On the motorcompressor with 60 Hz 4 pole motor a more noticeable improvement was obtained (about 18%) due to reduce inlet gas overheating obtained by the direct suction and due to the electric motor operation at temperature conditions for the optimal efficiency.

Adding the oil cooler the EER increase was of about 23%

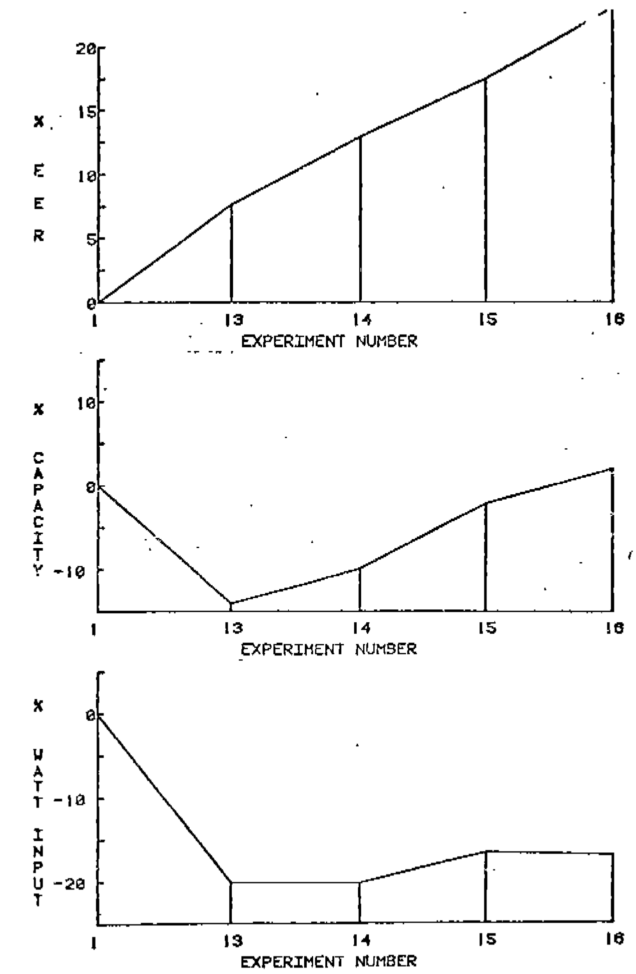

fig.6 - RL.4 - Percentage diagram of results

CONCLUSIONS

- The research for the best selection of the critical parts (valve plate, valve leaves, etc.) shows marginal EER improvements but it satisfactorily confirms the validity of design and accuracy already obtained in the compressor for the above parts.(RL.1)

- Not necessarily an increase of the 'clearance volume' ,due to an increase of the discharge ports section ,brings to an EER reduction because there is a minor loss of gas lamination through the ports.(RL.1)

- The electric motor speed reduction and displacement increase(at equal refrigerating capacity) has not given the expected improvements on the basis of a significant loss of lamination through the valves,due to the lower speed of the piston.(RL.2)

- The internal mechanical and electrical losses and heat transfer from piping and hot parts affect negatively the performance of a hermetic compressor because they increase the overheating of the gas at cylinder suction.(RL.2) and (RL.4)

- Direct gas suction has given major EER improvement to the other tested modifications. (RL.3) and (RL.4)

- Joining the revolutions speed reduction and direct gas suction connected to an electric 4 pole motor operating in optional efficiency conditions,we have obtained the 20% EER increase established as the goal of this research work.(RL.4)

- The compressor requires oil and liquid accumulators which are normally used in heat

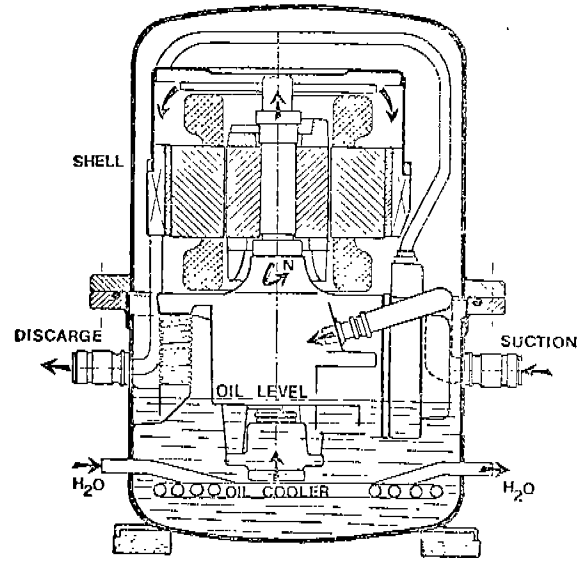

fig. 7 - *Schematic view of compressor used in RL.3 and RL.4*

pump system. Considering the extra costs involved in properly modifying the mechanic and electric parts, the economical results are encouraging but still to be improved.

APPENDIX

The correct manner to approach the internal heat transfer is that of using time dependent equations to describe the phenomena.

For the calculation the matter is not suitable because the time intervals of integration steps of such equations are not comparable with the brief ones which are used in the simulation of the fluodynamic gas behavior in the suction and discharge ducts.

It's instead suitable to solve the problem supposing the system at steady conditions and imposing an energy balance on the average values of the involved quantities.

It's evident that due to the complexity of the phenomena which are present, some simplifying assumptions should be made.

With the hypothesis that the equilibrium conditions are the same, the thermodynamic simulation provides :

$\dot{m}(N,T_C)$ = mass flow rate

$W_i(N)$ = crankshaft power

being:

N = rotation speed
T_C = cylinder suction temperature

and electric calculation motor program :

$W_O(N,\overline{T}_M)$ = output motor power

$W_D(N,\overline{T}_M)$ = motor losses

with:

$\overline{T}_M$ = average motor temperature

If we assume that the gas enters compressor shell at T_S temperature and is heated at a T_{F1} because of the W_M losses not owing to the electric motor, we could write :

$$(1) \qquad c_p\, \dot{m}\,(N,T_C)\,(T_{F1} - T_S) = K_1\, W_M$$

and subsequently:

$$(2) \qquad c_p\, \dot{m}\,(N,T_C)\,(T_{F2} - T_{F1}) = K_2\, W_D(N,\overline{T}_M)$$

$$(3) \qquad c_p\, \dot{m}\,(N,T_C)\,(T_C - T_{F2}) = K_3\, W_M$$

where K_2 and $(K_1 + K_3)$ are smaller than the unit to take in account the heat transfer to the ambient through the shell (see fig.8).

Motor losses are transferred to the working fluid and subsequently its temperature increases.

With the simplifying hypothesis that the electric motor-gas transfers depend on the average gas temperature

$$T_{gm} = (T_{F1} + T_{F2}) / 2$$

we could write :

$$(4) \qquad W_D(N,\overline{T}_M) = K_4 \left[\overline{T}_M - (T_{F1} + T_{F2}) / 2\right]$$

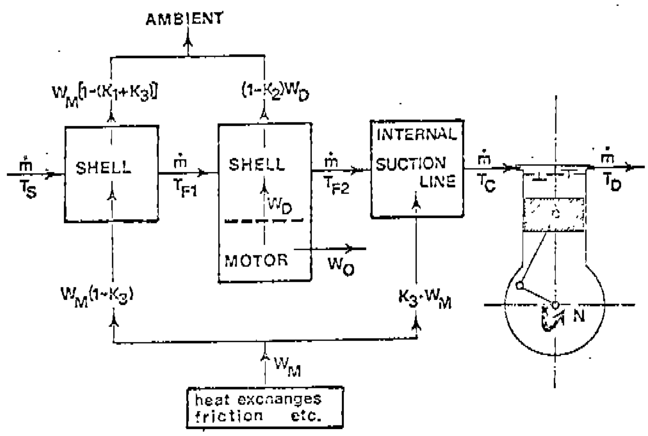

fig.7 - Block diagram for energy balance

The last equation which is to be written and which connects the variables is that which makes the motor output power equal to the power required by the thermodynamic cycle.

Under steady conditions, with the hypothesis that gas overheating has no influence on the claimed thermodynamic input power, such equation is :

$$(5) \qquad \eta_m\, W_O(N,\overline{T}_M) = W_i\,(N)$$

with η_m = mechanical efficiency

The solution of the previous equations is obtained analitically with the computer by successive iterations.

Such process can be evidenced grafically as shown in diagram of fig.9. The curves are traced taking a discrete number of points.

Solution (5) is reported in curve of Quadrant I. Therefore, a biunivocal correspondence between N and $\overline{T}_M$ is defined. Thus it is possible to draw the diagram of Quadrant II representing motor Watts losses (W_D) versus variations of the number of revolutions per minute.

With such a figure the system given by the formulas (1), (2) and (3) can be solved in a repetitive manner; the convergence points allow to diagram m in function of W (Quadrant III).

The system formed by (2) and by (4) is solved by obtaining diagram of Quadrant IV.

This last step has the meaning of using (4) as a disequation of power involved : the motor heats or cools according to mass gas flow rate which surrounds it and is able or not to dissipate lost power.

Following clockwise the diagrams from whatever conditions one starts one converges on a route obtaining : N = number of revolutions per minute , W_D = motor losses , $\dot{m}$ = mass flow rate and T_M = average motor temperature.

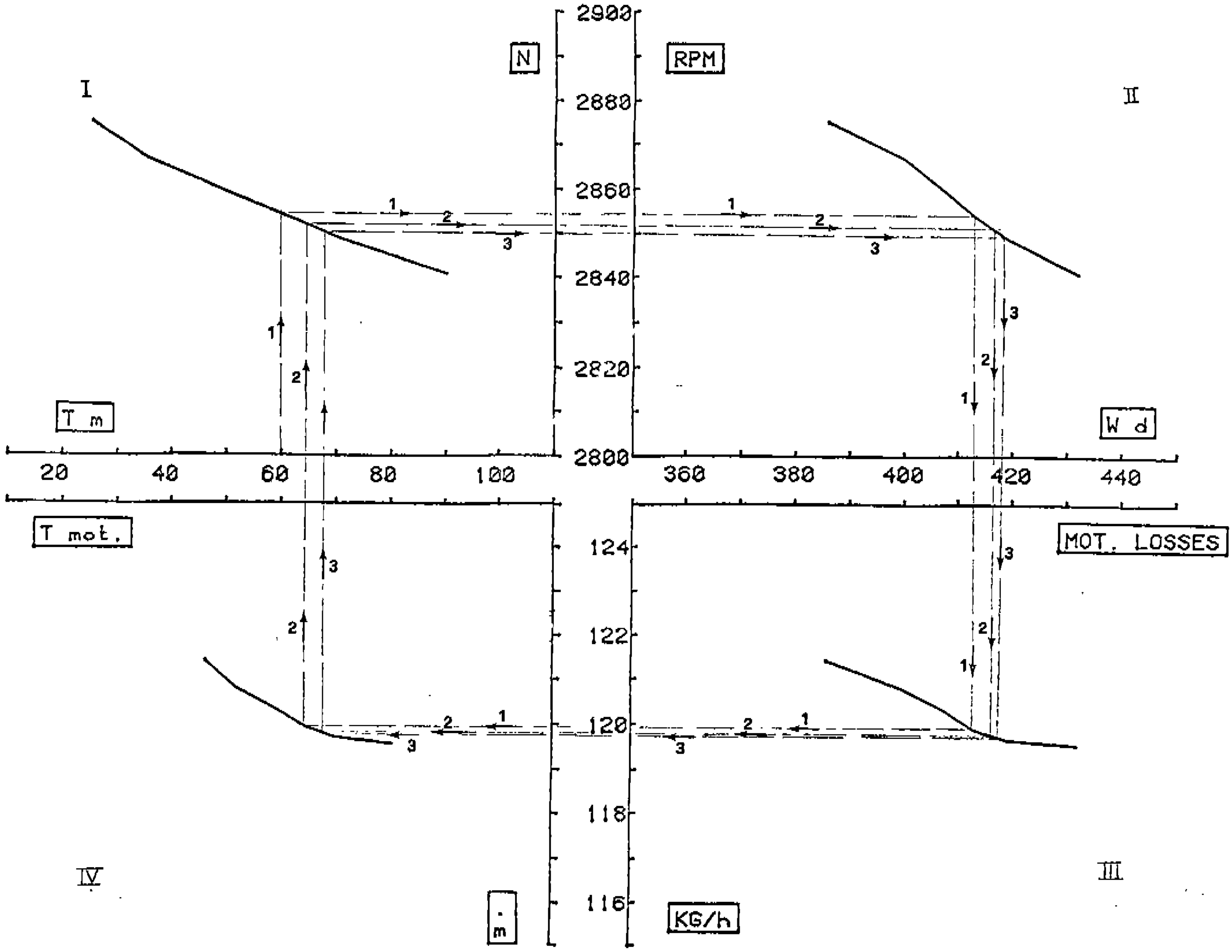

fig. 9 - Graphical resolution for original compressor energy balance

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